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CALCULATION OF SEDIMENT FORMATION ON FINNED HEAT EXCHANGER TUBES ON CONDENSATION OF A VAPOR–GAS MIXTURE WITH SOLID PARTICLES

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A mathematical model of the process of deposition of solid particles on the outer surface of tubes of a finned heat exchanger on condensation of a vapor–gas mixture containing particles has been developed. The model is based on determining the temperature field in a round fin from a given distribution of sediment thickness. The proposed model makes it possible to evaluate the influence of design and thermal parameters on the heat flux removed by a round fin to a cylindrical tube. The influence of changes in such parameters as the fin thickness and thermal conductivity of the sediment layer on heat transfer was assessed.

Keywords: *finned tubes, round fin, sediment formation.*

Introduction. Heat exchangers have found wide application in many industries thanks to processes such as cooling, evaporation, absorption, crystallization, etc. Use of a heat exchanger under difficult conditions certainly leads to its contamination. Often, this occurs due to the deposition of solid particles and fouling of the heat exchange surface. Industries that face the problem of constant fouling are the food industry, water treatment, pulp and paper production, fiber production, traditional and nuclear power plants, oil and gas refineries [1–4]. The development of effective methods for calculating heat and mass transfer processes is an urgent task in the field of energy efficiency and energy saving by thermal equipment, and it is considered by many researchers [5–8].

Despite the existing traditional and modern methods of minimizing deposits on the heat exchange surface, it is impossible to completely avoid contamination of heat exchangers, and deposits contribute to a decrease in the rate of heat transfer and to an increase in hydraulic resistance [9], which ultimately leads to an increase in capital costs and the costs of equipment maintenance, as well as to large production and energy losses [10, 11].

Today, researchers propose to use improved heat exchangers [12, 13]. Methods of heat transfer intensification in them have been studied to varying degrees, and only some of them have been brought to the level of industrial application [14]. A very effective means is external finning of the working heat transfer surface, but there are very few studies of deposits on such surfaces.

As a rule, the most problematic deposits are formed during the processing of solid or liquid waste and fuel combustion [15, 16]. In the case of using finned tubes for a heat exchanger, where the flue gases contain a lot of solid particles, the surface of the fins and tubes are contaminated quickly, and heat transfer is reduced as the hydraulic resistance increases. The influence of the design features of a finned heat exchanger on deposition of solid particles is determined by the pitch and number of rows of tubes [17, 18], pitch and type of fins [19, 20]. Experimental and numerical studies to determine the influence of particle size and inlet flow velocity on the deposition process show that the majority of particles are deposited on the front side of the heat exchanger [21]. Numerical simulation demonstrates that at the moment of deposition, large particles cannot pass between the fins of the heat exchanger and are retained by the formed layer on the front part of the heat exchange surface. It was established during experimental studies that introducing a large number of particles and increasing their sizes lead to an increase in the hydraulic resistance of the heat exchanger.

In [22], a numerical model was developed to predict the rate of particle deposition on the surface of the heat exchanger fins. The authors of the work modeled the process of solid particle deposition, which includes the transfer of

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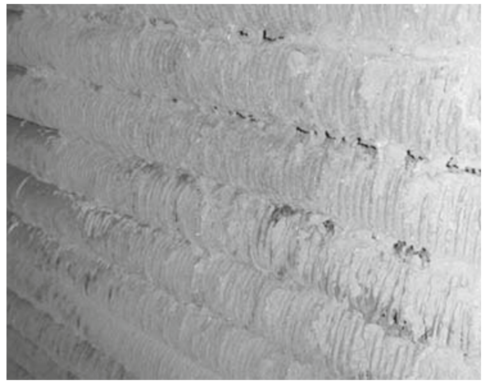

Fig. 1. Example of contamination of heat-exchanger tubes by sedimentation of solid particles: photo of the first row of finned-heat exchanger tubes with a sediment layer.

particles in the air, the impact of particles on the walls of the heat exchanger and on the layer of contamination on the fin surface.

In study [23], it was also found that the majority of particles are deposited mainly on the front part of the fins and tubes, and the minority on the leeward side, with the particles deposited on the fins forming a contaminant layer with lumps of contaminants being formed on the tubes. Also, the experimental data obtained showed that with a decrease in the fin pitch, the density of deposited particles increases at the same cross-sectional area of the channel, which increases the likehood of the collision of particles with the heat-exchanger surface, thereby increasing the volume of contaminants. Work [24] studied the influence of contaminants on the characteristics of the wettability of the heat transfer surface, which plays an important role in liquid–vapor phase transition phenomena, including the onset of boiling, critical heat flux, Leidenfrost transition, and condensation. It has been established that the wetting of heat transfer surface decreases with an increase in the thickness of the layer of contaminants.

Thus, understanding the mechanism of deposition on finned tubes of heat exchange equipment can contribute to controlling the fouling process [25] and therefore, to reducing its negative consequences for heat transfer equipment — reducing heat transfer and increasing hydraulic resistance, which accounts for the importance of this study.

The vapor–gas mixture consists of the products of combustion of technical gas, including steam and cellulose particles, which contributes to contamination and fouling of tubes. It can be seen from Fig. 1 that the fouling of tubes with transverse finning occurs evently. This is due to the fact that particles move to the surface due to steam condensation. Because of this, the profile of the heat flux to a contaminated fin differs significantly from the profile of the heat flux to a clean fin. The heat flux to the section of the fin, where the sediment thickness is greater, decreases, consequently, a more intense growth of the sediment on a cleaner surface begins. All this contributes to the formation of sediment of almost uniform thickness.

The goal of this work is to develop a method for calculating the process of sediment formation on the outer surface of finned tubes of a heat exchanger on condensation of a vapor–gas mixture containing solid particles. The proposed calculation method will make it possible to evaluate the influence of such parameters as fin thickness and thermal conductivity of the sediment layer on the heat flux removed by the fin to the tube.

Formulation of the Problem and Its Solution. A thin round fin of diameter $2R_D$ is located on a cooled tube of diameter $2R_0$ so that the temperature at its base t_0 = const. Steam condensation occurs on both sides of the fin at a temperature t_s = const, and contaminants settle on its surface. It is necessary to construct a model of the process, in particular, to determine the distribution of sediment thickness and the heat flux removed by the fin to the tube. The process is asymmetric with respect to the tube axis, so that the characteristics are independent of the rotation, and symmetric with respect to the midplane of the fin.

Let us introduce the main simplifying assumptions:

1) the fin is considered thin, so the thermal resistance over the thickness can be neglected, and the temperature distribution in the fin itself is assumed to be one-dimensional in the form of $t = t(r, \tau)$;

2) the sediment forms a homogeneous layer, which is considered thin, so that its longitudinal thermal conductivity in the direction parallel to the fin surface can be neglected;

3) the temperature of the sediment on the outer boundary is equal to t_s, and at the place where it is in contact with the fin it is equal to the fin temperature $t = t(r, \tau)$;

4) the heat capacity of the material of the fin and sediment can be neglected, then we obtain that the density of the heat flux through the sediment layer is equal to

$$q(r, \tau) = \frac{\lambda_{sed}}{\delta(r, \tau)} (t_s - t(r, \tau)) ; \tag{1}$$

5) the thickness $\delta(r, \tau)$ of the sediment layer at the given location r of the fin is proportional to the total quantity of condensate formed in this place;

6) the rate of condensate formation at the location of the fin with coordinate r at this moment τ is proportional to the heat flux density $q(r, \tau)$, the steam temperature is close to the condensation temperature t_s, the slight difference can be taken into account by using the correction for the condensation heat;

7) the sought functions $t(r, \tau)$ and $\delta(r, \tau)$ are continuously differentiable with respect to r and τ, with $t(r, \tau)$ being twice continuously differentiable with respect to r, except for the moment of $\tau = 0$.

Let us introduce the excess temperature $\vartheta(r, \tau) = t_s - t(r, \tau) > 0$ and, correspondingly, $\vartheta_0 = t_s - t_0$. The heat flux Q through an arbitrary cylindrical surface r = const in the fin, towards the base, according to the Fourier law will be equal to

$$Q(r, \tau) = -2\pi\Lambda r \frac{\partial\vartheta(r, \tau)}{\partial r} , \tag{2}$$

where $\Lambda = \lambda_f\delta_f$ = const is the thermal conductivity of the fin.

According to expression (1), a heat flux $2q(r, \tau)dF$ enters the fin through its side surfaces between r and r + dr, i.e, the change in Q from r to $r + dr$ will amount to $dQ(r, \tau) = -2 \dfrac{\lambda_{sed}}{\delta(r, \tau)} \vartheta(r, \tau) 2\pi r dr$ or

$$\frac{\partial Q(r, \tau)}{\partial r} = -4\pi\lambda_{sed} r \frac{\vartheta(r, \tau)}{\delta(r, \tau)} . \tag{3}$$

Differentiating expression (2) and combining it with expression (3), we obtain

$$\frac{\partial^2\vartheta(r, \tau)}{\partial r^2} + \frac{1}{r}\frac{\partial\vartheta(r, \tau)}{\partial r} = \frac{2\lambda_{sed}}{\Lambda}\frac{\vartheta(r, \tau)}{\delta(r, \tau)} . \tag{4}$$

The boundary conditions for $\vartheta(r, \tau)$ are

$$\vartheta(r, \tau)|_{r=R_0} = \vartheta_0 , \quad \left.\frac{\partial\vartheta}{\partial r}\right|_{r=R_D} = 0 . \tag{5}$$

The second condition corresponds to the absence of the heat flux on the end surface of the fin, which is considered to be negligibly thin.

It can be noted that in expression (4) there are no derivatives with respect to time τ. The time is only included as an argument in the sought functions, and it can be considered a parameter that determines the form $\delta(r, \tau)$. If at the moment τ the function $\delta(r, \tau)$ is known, Eq. (4) with conditions (5) must be solved uniquely as an ordinary second-order differential equation.

In accordance with the above assumptions, at the location of the fin with coordinate r over the period of time from τ to $\tau + d\tau$, the thickness of the sediment will increase by $d\delta$, proportional to $q(r, \tau)$, i.e, $d\delta = kqd\tau$, whence, taking into account Eq. (1), we obtain

$$\frac{\partial\delta(r, \tau)}{\partial\tau} = k\lambda_{sed}\frac{\vartheta(r, \tau)}{\delta(r, \tau)} . \tag{6}$$

The coefficient k is expressed through the concentration of contaminants in the condensate, a fraction of the contaminants settling on the surface, sediment density, and heat of condensation.

As the initial condition for $\delta(r, \tau)$ we can take

$$\delta(r,\,0) = h_0 = \text{const} > 0\,, \quad R_0 \le r \le R_D\,. \tag{7}$$

The formulation of problem (4)–(7) will make sense also at $h_0 = 0$, although such a solution at $\tau = 0$ will have a specific feature. In particular, $Q_0 \to \infty$ will be at $\tau \to +0$. However, for calculations it is more convenient to take a small value of $h_0 > 0$.

Let us first consider Eqs. (4) and (5) separately. They can be represented as the problem of determining the temperature field ϑ in a fin according to a given sediment thickness δ. In this case, if the thicknesses are given, the solution is independent of τ, the distribution of ϑ will be the same as in the stationary case with sediment. The parameter τ in such a separate consideration plays the role of an identifier of the distributions of sediment thickness $\delta(r)$ and of temperatures $\vartheta(r)$, corresponding to each other.

Construction of the distribution $\vartheta(r)$ according to the given distribution $\delta(r)$ by solution (4), (5) can be understood as application of some operation $\Re$ to $\delta(r)$, i.e., to designate $\vartheta(r) = \Re(\delta(r))$. In this case, the value of $\delta(r)$ can be taken to be quite arbitrary, for example, nonmonotonic. In general, the operator $\Re$ can hardly be expressed in an explicit analytical form. Its numerical representation is discussed below.

If we return to the general problem (4)–(7), then substituting $\Re$ into Eq. (6), we obtain the formulation of the problem in the form of the equation

$$\frac{\partial\delta(r,\,\tau)}{\partial\tau} = P\,\frac{\Re(\delta(r,\,\tau))}{\delta(r,\,\tau)} \tag{8}$$

relative to one function $\delta(r, \tau)$, where $P = k\lambda_{\text{sed}} = \text{const}$ is the temperature coefficient of sediment growth. Together with the initial condition (7), we obtain the Cauchy problem, which is solved numerically, for example, by the Euler, Runge–Kutta, or similar methods.

When analyzing the problem, the following properties of the operator $\Re$ may turn to be useful in problem (4), (5) considered separately at the given distribution $\delta(r, \tau)$:

1. When $\delta = \text{const}$, a problem arises that is equivalent to the classical round fin problem with normal heat emission, the coefficient α of which is equal to the thermal conductivity $\lambda_{\text{sed}}/\delta = \text{const}$ in this formulation, The problem has an exact analytical solution [25]

$$\frac{\vartheta(r)}{\vartheta_0} = \frac{I_0(mr)K_1(mR_D) - I_1(mR_D)K_0(mr)}{I_0(mR_0)K_1(mR_D) - I_1(mR_D)K_0(mR_0)}\,, \tag{9}$$

where $m = \sqrt{\dfrac{2\lambda_{\text{sed}}}{\delta\Lambda}}$.

2. From this solution it follows that when $\delta \to 0$ and $\alpha \to \infty$, the heat flux Q_0, removed by the fin, will grow without bound $Q_0 \sim \sqrt{1/\delta}$. This does not mean, however, that in the general problem it is theoretically impossible to use the initial condition of a "clean" fin $\delta\big|_{\tau=0} = h_0 = 0$.
3. Problem (4), (5) has a unique solution. This follows from the theory of differential equations, since expression (4) is a linear ordinary differential equation of the 2nd order with two boundary conditions at the ends.
4. The distribution $\vartheta(r)$, understood as a result of $\vartheta(r) = \Re(\delta(r))$ at any $\delta(r)$ is monotonically decreasing positive, i.e., $\vartheta_0 > \vartheta(r_1) > \vartheta(r_2) > 0$ at $R_0 < r_1 < r_2 \le R_D$. This is clear from the physical meaning.
5. With an increase of the thickness $\delta(r)$ of deposits at least at any place the temperature $\vartheta(r)$ increases over the whole fin $R_0 < r \le R_D$, and the heat flux Q_0 of the fin decreases.

Let us consider the numerical implementation of the operator $\Re$ — calculation of the distribution $\vartheta(r)$ of temperatures in a round fin according to the given distribution of the deposit thickness $\delta(r)$. The algorithm is built similarly to the case of the classical fin with ordinary heat emission as that described in [25]. Let us divide the region $R_0 \le r \le R_D = R_1 + l$ into n elements, each of which can be represented in the form of a ring (for internal $r_i - \Delta r/2 < r < r_i + \Delta r/2$, where $i = 1, \ldots, n-2$) with equidistant nodes at the points $r_i = il/(n-1)$, $i = 0, 1, \ldots, n-1$ (at the distance $\Delta r = l/(n-1)$ from each other). Instead of the function $\vartheta(r)$, we will look for nodal values ϑ_i, considering that at this stage δ_i [nodal values of the function $\delta(r)$], to be known (for $i = 0, 1, \ldots, n-1$).

The energy balance for each internal element is $Q_i = Q_{i+1} + dQ_i$ ($i = 1, \ldots, n-2$) (Fig. 2). The flux Q_i of heat along the fin from the element i to i – 1 in accordance with the finite-difference representation of the Fourier law will be

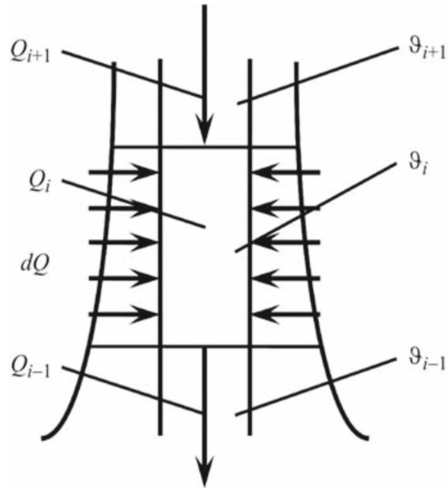

Fig. 2. Calculation diagram of heat fluxes.

$$Q_i = q_i F_i = \lambda_{\mathrm{f}} \frac{\vartheta_{i-1} - \vartheta_i}{\Delta r} 2\pi(r_i - \Delta r/2)\delta_{\mathrm{f}} , \tag{10}$$

where F_i is the area of the boundary between elements, q_i is the density of the heat flux through the boundary. Accordingly, the flux Q_{i+1} from the element $i + 1$ to i is equal to $Q_{i+1} = \Lambda \frac{\vartheta_i - \vartheta_{i+1}}{\Delta r} 2\pi(r_i + \Delta r/2)$. The heat flux dQ_i through sediment layers in the element i (from both sides) will be $dQ_i = 2q_\delta dF_i = 2\lambda_{\mathrm{sed}} 2\pi r_i \Delta r \vartheta_i/\delta_i$. After substitution we get

$$\left(1 - \frac{\Delta r}{2r_i}\right)\vartheta_{i-1} - 2\left(1 + \frac{\lambda_{\mathrm{sed}}}{\Lambda}\frac{\Delta r^2}{\delta_i}\right)\vartheta_i + \left(1 + \frac{\Delta r}{2r_i}\right)\vartheta_{i+1} = 0 , \quad i = 1, \dots , n - 2 . \tag{11}$$

An analogue of the boundary condition $\vartheta\big|_{r=R_0} = \vartheta_0$ (where $\vartheta_0 = \mathrm{const}$ is given) will be

$$\vartheta_{i=0} = \vartheta_0 . \tag{12}$$

On the boundary $r = r_{n-1} = R_D$ the balance condition $Q_{n-1} = dQ_{n-1}$ at $dQ_{n-1} = 2\lambda_{\mathrm{sed}} \frac{\vartheta_{n-1}}{\delta_{n-1}} \pi R_D \Delta r$ (taking into account the width $\Delta r/2$ of this element) takes the form

$$\left(1 - \frac{\Delta r}{2R_D}\right)\vartheta_{n-2} - \left(1 - \frac{\Delta r}{2R_D} + \frac{\lambda_{\mathrm{sed}}}{\Lambda}\frac{\Delta r^2}{\delta_{n-1}}\right)\vartheta_{n-1} = 0 . \tag{13}$$

It can be noted that here, when calculating dQ_{n-1}, the area dF_{n-1} is taken into account approximately as $\pi R_D \Delta r$ instead of the exact $\pi(R_D - \Delta r/4)\Delta r$; we can also indicate that ϑ_{n-1} is taken from the boundary rather than from the middle of the element, but the error here will be small because of the condition $\left.\frac{\partial\vartheta}{\partial r}\right|_{r=R_D} = 0$.

Thus, if δ_i and the remaining parameters are given, Eqs. (11)–(13) specify a system of n linear algebraic equations in n unknown ϑ_i. The matrix of the coefficients of the system is tridiagonal; it is solved by the sweep method.

Let δ_i^j and ϑ_i^j denote the nodal values of the sediment thickness and the temperatures at the nodes r_i at discreet moments τ^j. If for a moment τ^j ($j \geq 0$) the values of the thickness δ_i^j are known for all i ($0 \leq i \leq n - 1$), the algorithm outlined in the previous paragraph makes it possible to find the corresponding values of ϑ_i^j . To pass to the next moment τ^{j+1}, it is necessary to use a numerical analogue of Eq. (6) or (8), for example, using the Euler method:

$$\delta_i^{j+1} = \delta_i^j + P \frac{\vartheta_i^j}{\delta_i^j} \Delta\tau^j , \quad i = 0, \dots , n - 1 , \tag{14}$$

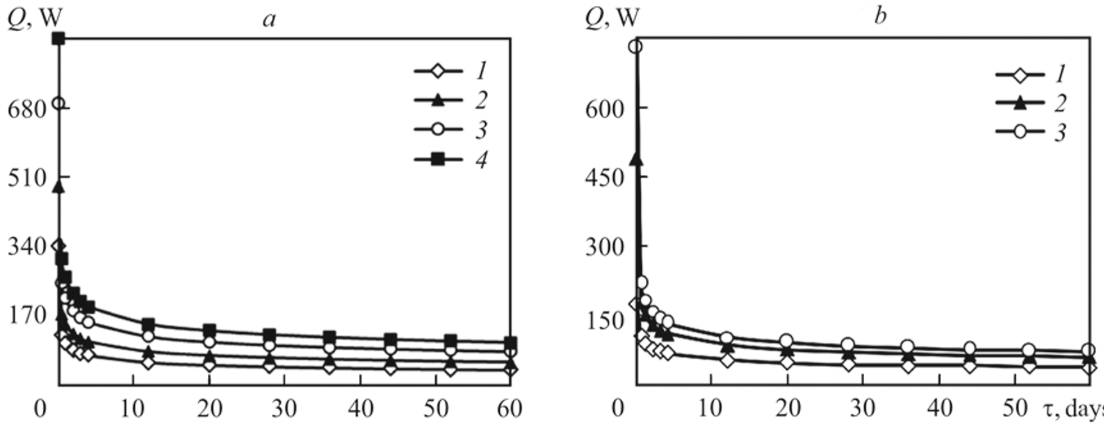

Fig. 3. Time dependences of the heat flux Q during 60 days of heat exchanger operation (a) at different fin thicknesses: 1) $\delta_f = 0.4$ mm; 2) 1.0; 3) 2.0; 4) 3.0; b) at different thermal conductivities of the sediment: 1) $\lambda_{sed} = 0.037$ W/(m·K); 2) 0.300; 3) 0.670.

where $\Delta\tau^j = \tau^{j+1} - \tau^j$, $j \geq 0$. The steps $\Delta\tau^j$ can be changed during the counting; at the initial stage, when the changes are quick and sudden, a small step should be taken for greater accuracy.

The numerical model is based on algorithm (12), (13), which implements the operator $\Re$. To estimate the efficiency and accuracy of this implementation, it is possible to use the above outlined property (1) of the operator $\Re(\delta)$, according to which in the case of δ = const it has an exact analytical representation (8). Calculations show that the error of numerical implementation in comparison with the analytical solution, for example, at $R_0 = 1$, $R_D = 2$, and $m = 1$ with the number of nodes $n > 200$ does not exceed 10^{-6}.

To check the accuracy of calculation of the entire process of sediment growth over time from finite-different relations (11)–(14), we can use property 2 of the solution of the complete problem, which gives an exact analytical solution (9) for the sediment thickness $\delta_0(\tau)$ at $r = R_0$. In this place, the sediment grows most quickly and nonuniformly, especially at the initial moments, and the appearance of the largest error can be expected. Calculations show that when a sufficient number of nodes n and small steps $\Delta\tau^j$ are selected at the initial moments, the finite-difference approach for typical initial parameters gives the sediment thickness with an error of up to thousands of a percent.

Calculation and Analysis of Results. Based on the developed model, the heat transfer apparatus with a finned outer surface of tubes was calculated. Figure 3 presents the results of assessing the influence of design and thermophysical parameters on the change of the heat flux removed by a round fin to the heat exchanger tube during condensation of a vapor–gas mixture with solid particles. The basic calculation parameters were: $R_D = 24.5$ mm, $\delta_f = 1$ mm, $R_0 = 11.45$ mm, $\lambda_f = 30$ W/(m·K), $\lambda_{sed} = 0.3$ W/(m·K). Sixty days were chosen days were chosen as the estimated operating time of the equipment.

Figure 3a shows the dependence of the heat flux Q removed by the fin to a tube on the time when the fin thickness δ_f changes from 0.4 to 3.0 mm. It can be seen that in 14.4 h an inflection point can be traced for all fins, after which the change in the heat flux up to 60 h of operation does not exceed an average of 35% for each fin thickness. It was revealed that when the thickness of the round fin is reduced from 3.0 to 0.4 mm, the heat flux decreases by 2.67 times at $\tau = 60$ h.

Thermal conductivity significantly affects heat transfer, therefore, further we studied the change in the heat flux on the change in the thermal conductivity of the sediment layer formed on the heat transfer surface.

Figure 3b shows the dependence of the heat flux Q removed by the fin to the tube during 60 days of heat exchanger operation when the thermal conductivity of the sediment changes. Here, the rate of decrease in Q is not the same at different thermal conductivities of the sediment. In particular, the decrease was equal to 73.2, 67.1, and 45.7% at the thermal conductivities of the sediment equal to 0.670, 0.300, and 0.037 W/(m·K), respectively. After 14.4 hours, the decrease in the heat flux is on average 26% at different thermal conductivities of the sediment λ_{sed}. An increase in the thermal conductivity coefficient of the sediment layer from 0.037 to 0.670 W/(m^2·K) leads to an increase in the heat flux by almost two times at time $\tau = 60$ h.

Thus, the significant decrease in the heat flux removed by the fin to the tube is observed at the beginning of the heat exchanger operation. This is associated with the formation of the first layer of sediment over the entire "clean" surface of the finned tubes, which leads to a rapid decrease in the heat transfer coefficient. The inflection point was observed at 14.4 h, after which a change in the heat flux is observed by no more than 25–35% depending on the design and thermophysical parameters under study. This indicates that heat removal occurs through a uniform layer of sediment, and its thickness proportionally affects the change in Q.

Conclusions. The paper proposes a method for calculating the process of sediment formation on the outer surface of finned tubes of a heat exchanger during condensation of a vapor–gas mixture containing solid particles. It was found that during operation for 60 days the heat flux decreases by 2.67 times if the thickness of fins is reduced from 3 to 0.4 mm. The thermal conductivity of the sediment layer significantly influences the process. An increase in the thermal conductivity of the sediment layer from 0.037 to 0.670 W/(m·K) leads to an almost two-fold increase in the heat flux.

The proposed calculation method makes it possible to evaluate the contribution of thermophysical parameters and structural dimensions of a finned tubular heat exchanger. It is worth noting that the calculation of the entire simulated process of the growth of deposits on finned tubes of heat exchanging equipment over time can be carried out using the developed method for the case of a straight fin.

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NOTATION

h_0, initial thickness of sediment, m; I_0, I_1, K_0, and K_1, Bessel functions; k, thermal coefficient of sediment increment; n, number of elements, pcs; P, temperature coefficient of sediment growth; $Q_0(r, \tau)$, heat flux along the fin, W; Q_i, heat flux along the fin from element i to $i - 1$, W; Q_n, heat flux removed from n elements, W; $q(r, \tau)$, density of heat flux through the deposit layer, W/m^2; $2R_D$, diameter of the round fin, m; R_0, radius of the cylindrical tube, m; r, radial coordinate of the fin; t_0, temperature at the base of the fin, $^{\circ}$C; t_s, condensation temperature, $^{\circ}$C; $t(r, \tau)$, temperature of the fin, $^{\circ}$C; α, heat transfer coefficient, W/(m^2·$^{\circ}$C); δ_f, fin thickness, m; $\delta(r, \tau)$, thickness of sediment layer; $\vartheta(r, \tau)$, excess temperature, $^{\circ}$C; $\vartheta(r)$, temperature distribution in a round fin; $\lambda_{sed}/\delta(r, \tau)$, thermal conductivity of the sediment; λ_f, thermal conductivity of the fin, W/(m·$^{\circ}$C); λ_{sed}, thermal conductivity of sediment material, W/(m·$^{\circ}$C); τ, time, s; $\Re$, operator. Indices: f, fin; sed, sediment.

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